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Performance of Two Plate-Type Ice-Maker Heat Pumps*

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ABSTRACT

An ice-maker heat pump is a device that extracts heat from a fixed volume of water to provide space and water heating. The ice formed is periodically harvested and stored in a water/ice storage bin for later use in space cooling.

A Remcor ice-maker heat pump utilizing compressor discharge gas for harvesting and a two-plate unit with a "no-penalty" harvest scheme were tested. The Remcor's harvest scheme exacted a heavy penalty on performance during short freeze cycles as compared to the no-penalty scheme. However, its lower plate loading gave it a better coefficient of performance than the two-plate unit on long freeze cycles and while chilling water. Cyclic operation of the two-plate unit revealed the possibility of improving ice-maker performance and reliability by combining duty cycles with freeze-harvest cycles.

THE SHARP INCREASE in the price of energy since 1973 has revived interest in a number of old, but valid, ideas of how to utilize energy in more efficient ways. One of these is the Annual Cycle Energy System (ACES),(1,2)** an integrated system which uses a heat pump and energy storage to provide space heating, space cooling, and domestic hot water. During the heating season, the heat pump extracts the required heat from a tank of water, converting a portion of the water to ice, which is stored to provide cooling during the summer. For this system, since both the heating and cooling outputs of the heat pump are used simultaneously, the annual efficiency is considerably higher than for conventional systems.

In the original conception, a chilled antifreeze solution is circulated through tubing submerged in the water tank, and ice freezes on the tube surface. An alternative, probably less expensive approach would utilize an ice-maker heat pump (IMHP). An IMHP would have the ice formation occurring on refrigerated plates located above the ice bin. Periodically, the ice would be harvested and dropped into the bin. This approach has the advantage of modular construction of the entire unit — exclusive of the ice bin — by the manufacturer, thus obviating the expense of field construction of the submerged tubing heat exchanger.

The purpose of this paper is to compare the performance of two IMHPs, the two-plate unit developed by Fischer(3) and a unit built to specifications for test purposes by Remcor Products Co. There are two major differences between the two machines. The first is the type of harvesting scheme employed. The two-plate machine uses warm refrigerant vapor from a receiver for harvesting, resulting in a "no-penalty" process in which no condenser output is lost due to the harvest. In the Remcor system, hot gas from the compressor is used, resulting in no heat output from the condenser during harvest. The second difference is due to the evaporator and compressor size. There are four plates in the Remcor with a total active area of 3.51 m^2, while the two-plate unit has an active area of 2.79 m^2. Also, because the Remcor has a smaller compressor, its capacity as tested is approximately two-thirds that of the two-plate unit, resulting in an evaporator plate loading on the Remcor of 60 to 70% of that on the two-plate unit. The tests discussed in this paper were performed to show the effect of these two differences on the relative performance of the machines in both water-chilling and ice-making modes.

In addition to the tests mentioned above, the effect of cyclic operation on performance was investigated.

DESCRIPTION OF TEST UNITS

REMCOR IMHP-The Remcor unit, shown schematically in Fig. 1, was mounted atop a large water storage tank in the laboratory. A water pump delivered the tank water to a header arrangement which distributed the water over the evaporator plates during operation. Originally, the unit was equipped with a compressor having a rated heating capacity of ~8793 W for a condenser temperature of 40.6°C and an evaporator temperature of −3.9°C. During preliminary testing, this compressor was found to be defective and was replaced. The heating capacity of the unit with the new compressor at 40.6°C condensing and −6.7 to −3.9°C evaporating is ~5862 W.

The evaporator consisted of four copper plates having an overall area of 4,65 m^2. In operation, water covered an area of 3.51 m^2. Refrigerant circuitry through the plates consisted of a 12.7-mm tube making 14 passes on 50.8-mm tube centers.

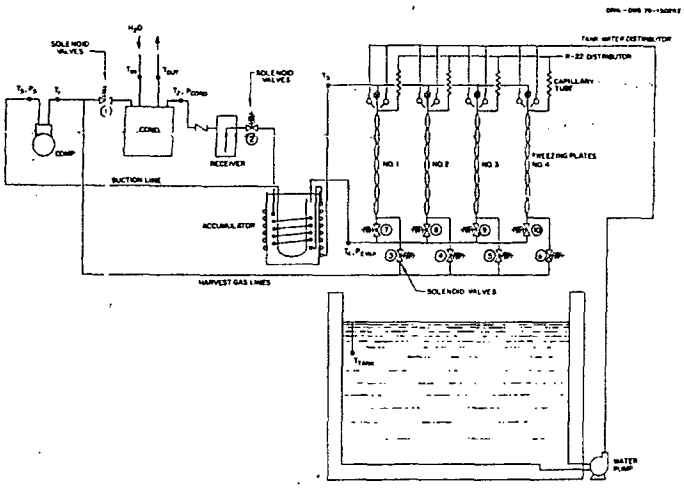

Fig. 1 - Remcor schematic diagram.

Under normal conditions of chilling water or making ice, the unit operates with solenoid valves 1, 2, and 7–10 open and 3–6 closed allowing refrigerant to circulate from compressor to condenser and evaporator and back to the compressor. When the harvest cycle begins, valves 1 and 2 close, bypassing the condenser and thereby reducing the heating output to zero. The plates are harvested in order 1, 2, 3, and 4. For plate 1, valve 7 closes and valve 3 opens, admitting hot gas from the compressor to the bottom of the plate. As the hot gas moves up through the plate, it condenses, thereby warming the plate, melting the ice at the interface, and causing the ice to drop into the tank. The condensed R-22 exits the plate through the capillary tube to the R-22 distributor. It then flows through the capillary tubes into the other plates, where it evaporates and returns to the compressor through the suction line. Plates 2–4 are harvested in a similar manner. After the harvest cycle is complete, valves 1 and 2 reopen and normal operation resumes. Total time for harvest was approximately 3.5 min. The total length of the freeze-harvest cycle was controlled by a timing system that allowed a maximum cycle time of 73.5 min.

TWO-PLATE IMHP-The two-plate unit, shown in Fig. 2, was mounted atop a second water tank in the laboratory. Its heating capacity was about 7325 W under the same conditions as above.

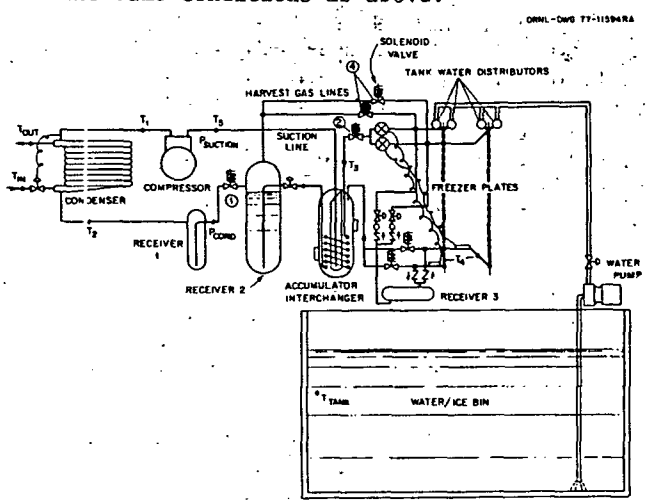

Fig. 2 - Two-plate schematic diagram.

The evaporator of this unit consisted of two mild-steel plates having a total active surface area of 2.79 m^2. Refrigerant circuitry was formed by welds on 38.1-mm centers with 12 passes per plate.

In normal operation, solenoid valves 1, 2, and 3 are open, and the evaporator plates are freezing ice or chilling water, depending on the tank temperature. Refrigerant is circulating through the condenser, receivers 1 and 2, accumulator, evaporator plates and back to the compressor suction with the excess charge filling up receiver 2. To effect harvest, valves 1 and 2 close and one of the valves 3 closes. The corresponding harvest line valve 4 opens connecting the plate to be harvested (at the suction side pressure) to the gas space at the top of receiver 2 (at the high side pressure). The liquid in the receiver immediately begins boiling, forcing the warm vapor into the top of the harvesting plate. This vapor condenses in the plate, warming it and causing the ice to drop into the bin. The condensed liquid is then routed through receiver 3 to the top of the other plate, where it evaporates to freeze ice and returns to the compressor via the accumulator.

The freeze-harvest cycle of the two-plate machine has been described by Fischer(3) as follows:

1. both plates freeze (typical time, 7.5 min);
2. first plate harvests while the second plate continues to freeze (typical time, 35 sec);
3. both plates freeze (typical time, 7.5 min);
4. second plate harvests while the first plate continues to freeze (typical time, 35 sec);
5. both plates freeze (typical time, 7.5 min).

The length of the freeze and harvest times was controlled by a timer with a maximum of 20.5 min.

TEST APPARATUS AND INSTRUMENTATION

The heat pump heating capacity was measured by observing the condenser cooling water temperature difference, using copper-constantan thermocouples, and monitoring the total coolant flow by weighing (for the Remcor) or with a water meter (for the two-plate). Cooling capacity was not directly measured in these tests. It could, however, be estimated by performing a heat balance on the refrigerant circuit.

Electrical input to the compressor was measured by a thermal-watt converter. In addition, a watt-hour meter was used as a backup and as a check on the primary instrument.

Refrigerant cycle temperatures and pressures were measured by copper-constantan thermocouples and pressure gages located as indicated in Figs. 1 and 2. These measurements were taken as a check on cycle operation and for use in computing refrigerant-side heat balances.

All temperatures and the compressor electrical input were recorded and averaged by a data acquisition system. This system consisted primarily of an analog-to-digital converter, tape reader, Teletype, and scanner. Calibration data entered into the computer were used to convert digital voltage inputs to the desired units for temperature and power input. The computer recorded temperatures and compressor power input every 60 sec on paper and on

paper tape. Averaging of the data was accomplished by using the paper tape, tape reader, and minicomputer.

TEST PROCEDURE

The procedure for water-chilling tests for the Remcor unit was as follows:

1. The unit was turned on and allowed to run for approximately one-half hour to establish steady-state conditions in the refrigerant circuit. Since harvesting is unnecessary while chilling water, the harvest cycle was bypassed during these tests.
2. Every 15 min, condenser water flow was weighed for a 3-min period, during which time refrigerant cycle pressures were recorded. Refrigerant temperature, condenser cooling water temperatures, tank water temperature, and compressor electrical input were recorded every minute and averaged over the test period.
3. Step two was repeated until the unit began to make ice.

For the two-plate machine, the test procedure was somewhat different. The unit was operated without harvest; however, the tests were of 1-hr duration, with temperatures and compressor input recorded every minute and averaged at the end of a test. The tank water temperature seldom fell more than 1.1°C during a test. Water flow was recorded by means of a flowmeter. Pressures were read from gages every 10 min.

For the ice-making tests, test duration was at least 1 hr, but never less than 2 full freeze-harvest cycles starting and ending at the end of a harvest. Temperatures and compressor power usage were obtained through digital readout from the data acquisition system each minute. Pressures were read from gages as before. Cooling water flow rate was measured via water meter for the two-plate machine and by weighing for the Remcor.

Prior to running cyclic tests, the two-plate machine was allowed to warm up for about 30 min; then, with a freeze cycle length of 12 min, a full-load steady-state test was run to provide a base point for comparison with cyclic test results. Each cyclic test was run for a 1-hr period, beginning and ending at the end of an on-cycle. Cycling rates of 2, 4, and 6 cycles per hour and compressor on-time ratios of 10, 20, 50, and 80% were used in the tests.

The coefficient of performance (COP) referred to subsequently in this paper is a compressor-only COP; that is, useful heat output divided by compressor electrical input.

WATER-CHILLING TEST RESULTS

Both units were tested in the water-chilling mode to determine the effect of evaporator temperature and plate loading on performance.

Test results indicate that heating capacity, COP, and evaporator temperature decreased as the tank temperature decreased for both machines. Comparing the results of the two units (Fig. 3) indicated that, for a given water temperature, the Remcor had a higher evaporator temperature and therefore a higher COP. By looking at the heat transfer equation for the plates,

$$Q_E = UA_p(T_{H_2O} - T_{R-22}) ,$$

and noting from Table 1 that the evaporator load on the Remcor is approximately 85% that on the two-plate machine, it is possible to determine why this is so. We can define a specific plate loading, $\overline{Q}_E$, as

$$\overline{Q}_E = Q_E/A_p = U(T_{H_2O} - T_{R-22}) ,$$

where

Q_E = evaporator load,
A_p = evaporator active area,
U = overall heat transfer coefficient between water and refrigerant,
T_{H_2O} = water temperature,
T_{R-22} = evaporating refrigerant temperature.

For these calculations, the evaporator load was calculated as follows:

$$Q_E = Q_o - F \cdot W_{in} ,$$

where

Q_o = condenser heat output,
W_{in} = compressor energy input,
F = fraction of electrical input to compressor that is added to refrigerant stream through compressor.

The factor F was found by performing heat balances on the refrigerant side of the systems. Since the condensers were well-insulated from the ambient air, and the surface temperature of the

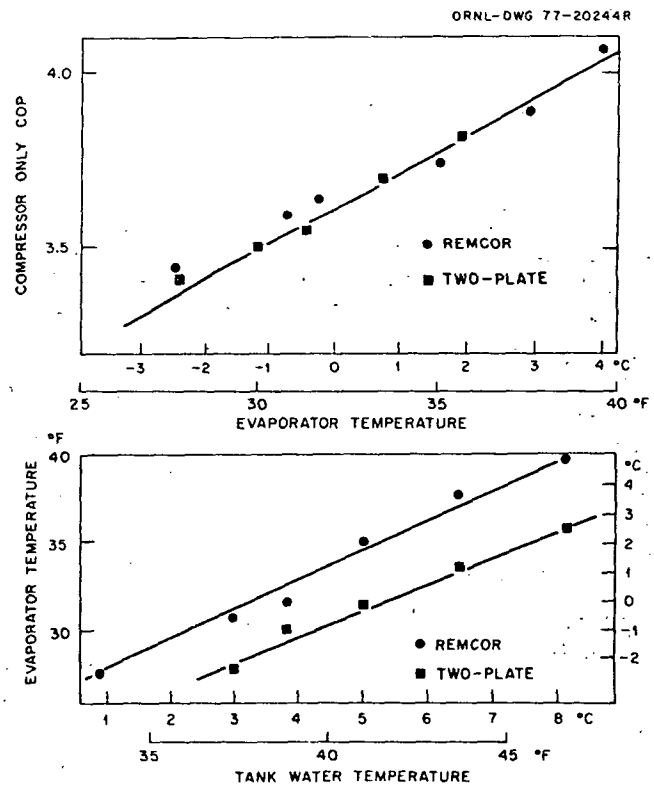

Fig. 3 - IMHP performance as a water chiller.

Table 1 - Water-Chilling Data from Tests on Two-Plate and Remcor IMHPs

Tank temperature, °F (°C)	46.7 (8.2)	43.7 (6.5)	41.0 (5.0)	38.9 (3.8)	37.4 (3.0)
Heating output, Btu/h (W)					
Two-plate	37,108 (10,876)	35,124 (10,294)	33,139 (9712)	32,079 (9402)	30,797 (9026)
REMCOR	28,720 (8417)	27,134 (7953)	25,749 (7547)	24,747 (7253)	24,165 (7082)
Compressor input, Btu/h (W)					
Two-plate	9721 (2849)	9504 (2786)	9334 (2736)	9180 (2691)	9042 (2650)
REMCOR	7080 (2075)	6984 (2047)	6866 (2012)	6771 (1984)	6704 (1965)
Evaporator load, Btu/h (W)					
Two-plate	28,539 (8364)	26,570 (7787)	24,738 (7250)	23,817 (6980)	22,654 (6441)
REMCOR	23,410 (6861)	21,896 (6417)	20,600 (6038)	19,669 (5765)	19,137 (5609)
Evaporator temperature, °F (°C)					
Two-plate	35.7 (2.1)	33.5 (0.8)	31.4 (−0.3)	30.0 (−1.1)	27.8 (−2.3)
REMCOR	39.6 (4.2)	37.6 (3.1)	35.1 (1.7)	31.7 (−0.2)	30.8 (−0.7)
Specific plate loading, Btu/h·ft^2 (W/m^2)					
Two-plate	951.3 (3000.0)	885.7 (2793.5)	824.6 (2600.8)	793.9 (2504.0)	755.3 (2382.2)
REMCOR	619.6 (1954.2)	579.6 (1828.1)	545.3 (1719.9)	520.6 (1642.0)	506.5 (1598.0)
U factor, Btu/h·ft^2·°F (W/m^2·°C)					
Two-plate	86.5 (491.2)	86.8 (492.8)	85.9 (487.7)	89.2 (506.5)	78.7 (446.9)
REMCOR	87.3 (495.7)	95.0 (539.4)	92.4 (524.6)	72.3 (410.5)	76.8 (436.1)
F factor, dimensionless					
Two-plate	0.91	0.90	0.91	0.89	0.87
REMCOR	0.74	0.76	0.75	0.73	0.75

insulation was only 2–3°C higher than the ambient air, it can be assumed that the heat loss to the surroundings is insignificant compared to the total heat transfer between water and refrigerant sides. For this case, the refrigerant flow rate can be determined by

$$\dot{m}_{R-22} = \frac{Q_o}{h_1 - h_2},$$

where the enthalpies h_1, h_2 are determined from the corresponding refrigerant temperatures and pressures taken at points shown in Figs. 1 and 2. The factor F was then calculated by

$$F = \frac{\dot{m}_{R-22}(h_1 - h_5)}{W_{in}}.$$

From analysis of several tests, average values of F were found to be 0.75 for the Remcor and 0.90 for the two-plate heat pump. These values were then used to calculate all the evaporator loads listed in Table 1.

As seen in Table 1, the specific plate loading factor was much smaller for the Remcor than for the two-plate machine. This is due to having both lower capacity and more evaporator area. The heat transfer coefficients for both machines, obtained by:

$$U = \frac{\overline{Q}_E}{T_{H_2O} - T_{R-22}},$$

are approximately the same. Therefore, the Remcor must have a lower temperature difference between water and refrigerant and thus a higher evaporator temperature.

ICE-MAKING TEST RESULTS

The performance of the two heat pumps while making ice was checked for the effect of two different parameters: (1) freezing cycle time and (2) condenser temperature. For both of these variables, the effects of plate loading and the two harvesting schemes were investigated.

Figure 4 illustrates the performance of both units as a function of freezing time with and without harvesting. The no-harvest curve for the two-plate unit was obtained experimentally by running the unit with the harvesting circuit bypassed. For the Remcor, this curve was obtained from the data taken during the freeze-harvest cycle tests by ignoring the data points taken during harvest. This is possible because all of the heating output of the Remcor occurs during the freeze portion of the cycle. Some data points for the no-harvest curves are omitted for clarity.

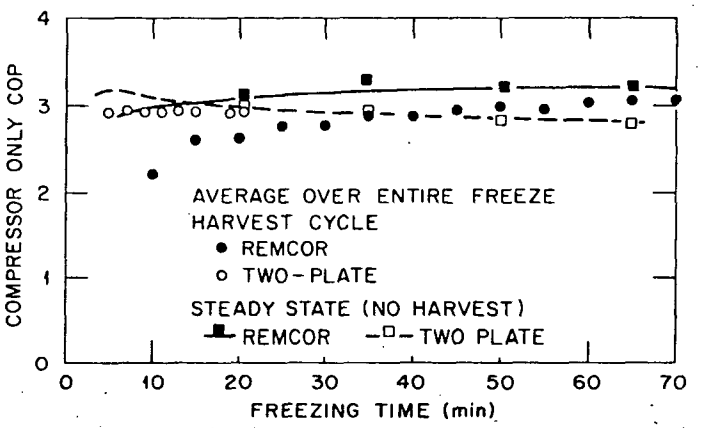

Fig. 4 - Compressor-only COP vs length of freeze between harvests.

As can be seen, for freezing times less than 20 min, the two-plate machine had an advantage when operating with the harvest cycle engaged. The reason is as follows: when the Remcor was harvesting, no useful heat was produced, that is, the unit operated with a heating COP of zero for the 3-1/2-min harvest. COPs without harvest for short freezing times were about the same for both machines.

As freezing time increased past 20 min, and ice thickness increased, the steady-state performance of the two-plate machine began to fall off while that of the Remcor continued to rise for both freeze-harvest cycle and steady-state operation. No freeze-harvest cycle tests for freeze times longer than 20 min were run on the two-plate machine, therefore it is impossible to say exactly

what effect longer freeze times would have had on its performance with harvesting. It is reasonable to assume, however, that freeze-harvest performance would follow the steady-state performance with slightly lower COPs.

There are two possible reasons for the improvement in the Remcor's performance with increasing freeze times:

1. The Remcor may have picked up more heat from the surrounding air than did the two-plate unit, thereby holding its evaporator at a higher temperature.
2. The lower plate loading of the Remcor retarded the buildup of ice on the plates, thus allowing it to run longer between harvests and negating the effect of the relatively short harvest period.

The data show that both machines achieved a peak COP in the ice-making mode of around 3.00. Apparently, a combination of the above two factors allowed the Remcor to overcome the deleterious effect of using compressor discharge gas for harvesting by enabling it to run very long freeze-harvest cycles.

Ambient air effects can be divided into two parts: (1) natural convection to the unwetted edges of the plates and (2) direct heat transfer from the air to the water flowing across the plates. The Remcor has about 2.5 times as much unwetted area as the two-plate unit. However, the natural-convection coefficients are on the order of 2.84 $W/m^2 \cdot K$, and therefore the natural-convection effect is considered negligible. Direct heat transfer to the water film probably has a much greater effect, which has not yet been quantified. Experiments with an insulated enclosure around the freezing plates of the Remcor indicated performance drops of about 1–2% and harvesting difficulty at freeze times of over 35 min. Minimum harvesting time required increased by 35–50% for freeze times of 35 min or less, and by 15% for a 70-min freeze time.

It must be said that warm air near the evaporator plates probably had some retarding effect on ice growth on both machines with a somewhat greater effect on the Remcor due to its smaller capacity and larger evaporator. The magnitude of this effect is not known; however, it is assumed to represent a relatively small fraction of the total heat gain by the evaporators of both units. On this basis it is believed that the primary reason for the Remcor's ability to operate with such long freeze times is its lower specific plate loading.

Experiments performed on the Remcor with reduced plate area bear out this assertion. Figure 5 shows heating capacity, COP, and evaporator temperature as functions of specific plate loading. These data were obtained by testing the Remcor with all four plates freezing, three plates freezing, and so on down to one plate freezing. These tests were of the steady-state, no-harvest type, performed with the insulated enclosure around the plates. Test duration was 30 min each. As the evaporator size decreased, the performance suffered.

Ice-maker performance as a function of condenser temperature is shown in Fig. 6. Both machines behaved in a similar manner, with COP and capacity rising as condenser temperature decreased.

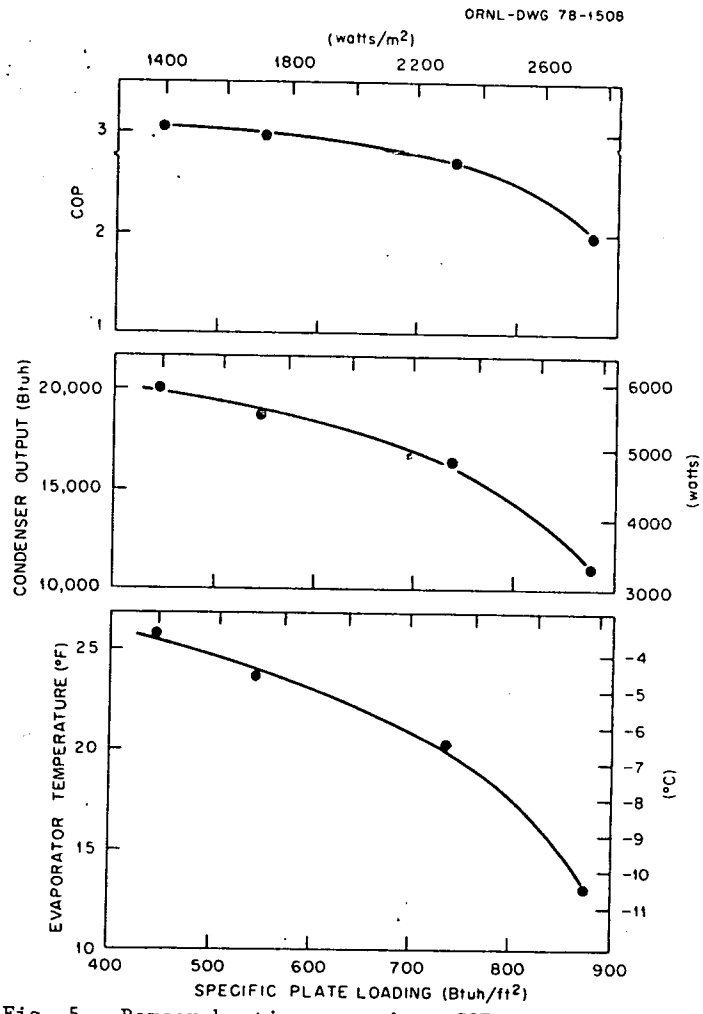

Fig. 5 - Remcor heating capacity, COP, and evaporator temperature as functions of specific plate loading.

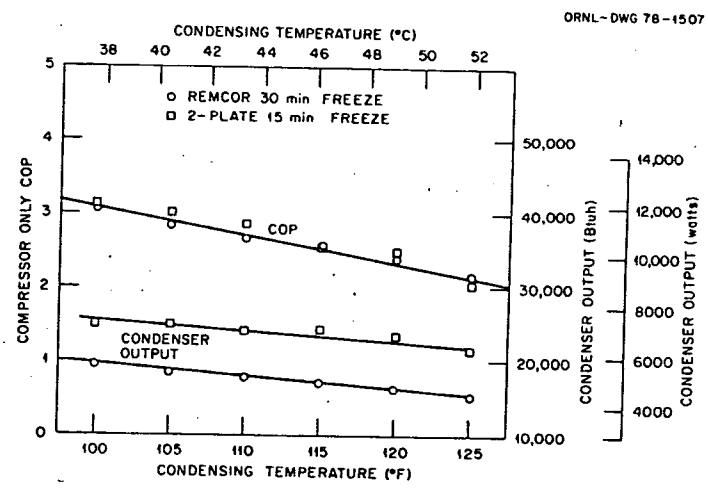

Fig. 6 - Two-plate IMHP performance as a function of condenser temperature.

As can be seen, the Remcor very closely matched the COP of the two-plate machine despite its use of hot gas from the compressor for harvest. This was possible because it could freeze ice for long

enough periods of time to negate the penalty involved in harvesting.

CYCLIC TEST RESULTS

The two-plate machine was tested under various cyclic conditions typical of the partial loading a heating system may see during the early and late parts of the heating season. Performance effects of part loading, compressor run time, and cycling rate were examined.

In order to present the data more effectively, a part-load output factor was defined as follows:

$$PF = Qo_c/(Qo_{ss} \times \Delta t_t) ,$$

where

Qo_c = total cyclic heat output, Whr,

Qo_{ss} = steady-state, full-load heat output rate, W,

Δt_t = total test duration, hr.

Test results are presented in Table 2 and in Figs. 7 through 9. The figures show the relationship of the ratio of cyclic COP to steady-state COP (COP_c/COP_{ss}) to the part-loading factor, compressor run time, and percent compressor on-time.

Table 2 - Cyclic Test Results on Two-Plate Machine

Comp. on time (%)	Comp. on time (min)	Qo_c (Btu)	Qo_c (Whr)	Cycles per hr	PF	COP_c/COP_{ss}
10	3	1,726.6	506	2	0.07	0.69
10	1.5	1,100.0	322.4	4	0.05	0.44
20	6	4,381.7	1284.2	2	0.18	0.86
20	3	3,162.2	926.8	4	0.13	0.63
50	15	11,754.5	3445.0	2	0.49	0.93
50	7.5	11,632.0	3409.1	4	0.48	0.91
50	5	11,051.5	3239.0	6	0.46	0.87
80	24	19,076.7	5591.1	2	0.79	0.94
80	12	18,867.9	5529.9	4	0.78	0.93
80	8	18,697.3	5479.9	6	0.77	0.93
100*		24,231.3	7101.8		1	1

*Full-load steady-state test.

For a part load of 10%, Fig. 7, the machine achieves 70% of its steady-state COP. From Fig. 8 and Table 2, it can be seen that this performance level is obtained for compressor run times of 3 min or longer. The effect of cycling rate can be seen in Fig. 9. For a given percent compressor on-time, performance improves as the cycling rate decreases.

It might be noted that as the load factor, PF, approaches 1.00, the ratio COP_c/COP_{ss} appears to approach some value less than 1.00. This can be accounted for because the test duration was limited to 1 hr. For longer test periods, allowing for longer on-cycle times, it is felt that the COP ratio would indeed approach 1.00.

The losses incurred due to cyclic operation of the two-plate machine with its no-penalty harvest leads to an interesting observation. Since IMHP operation is cyclic by definition, and typical field operation under thermostat control is also cyclic, it should be possible to combine the cycles such that the ice is harvested simply due to the machine being turned off, thus obviating the need for an elaborate harvest scheme. If such a

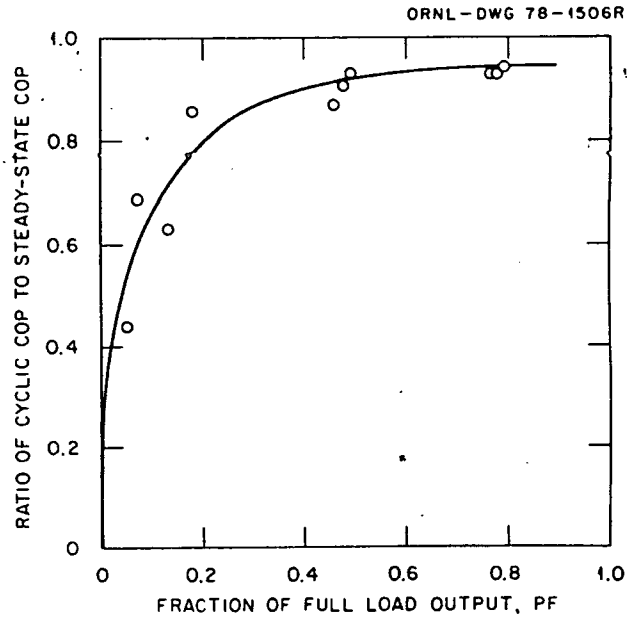

Fig. 7 - Two-plate IMHP, ratio of cyclic COP to steady-state COP to part-loading factor.

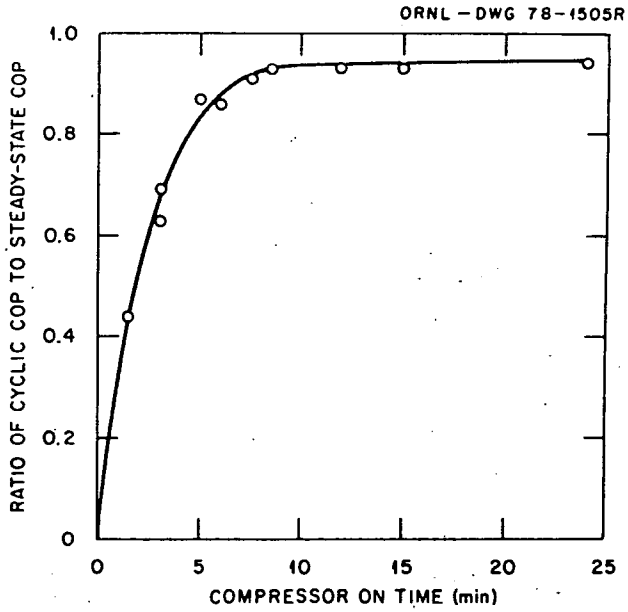

Fig. 8 - Two plate IMHP, ratio of cyclic COP to steady-state COP to compressor run time.

machine had a low enough plate loading level to allow it to run for 75 to 90 min before losing on performance, then operation need never be interrupted for harvesting except on those very cold days when the thermostat requires the unit to run continuously. For such occasions, a timer can be incorporated to turn the unit off for approximately 5 min for harvest, after which it could resume operation.

SUMMARY

WATER-CHILLING TEST RESULTS AND CONCLUSIONS- Both units exhibited the same tendencies, in that heating capacity, COP, and evaporator temperature

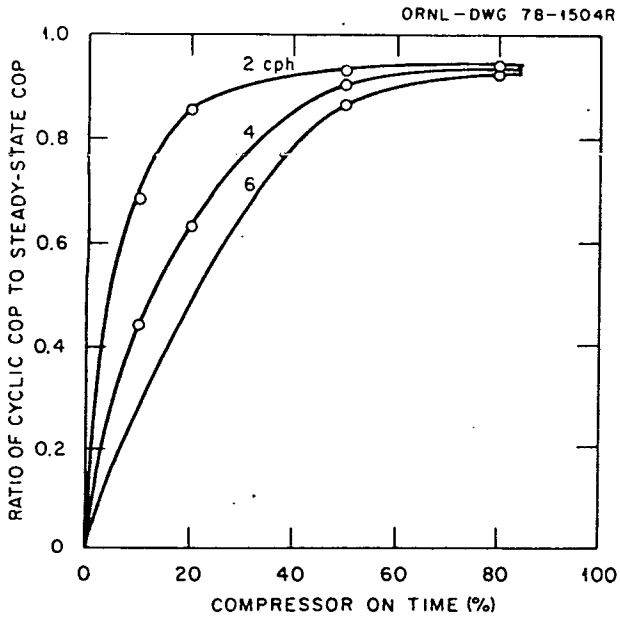

Fig. 9 - Two-plate IMHP, ratio of cyclic COP to steady-state COP to percent compressor on-time.

decreased as tank water temperature decreased. However, for a given tank water temperature, the Remcor COP was higher. For example, at a tank temperature of 8.2°C, it had a COP of 4.06 while the two-plate machine had a COP of 3.82. This effect is attributed to lower specific plate loading on the Remcor causing it to operate at a higher evaporator temperature.

ICE-MAKING TEST RESULTS AND CONCLUSIONS-The Remcor's system of using compressor discharge gas for harvesting ice caused its condenser output to drop to zero during the harvest. This resulted in the machine suffering a severe performance penalty when operating on short freeze cycle times (20 min or less), giving COPs of about 2.2–2.7. The two-plate unit, due to its no-penalty harvesting scheme, experienced no such penalty and had a COP of around 2.94 for those freeze cycle times.

For freeze times longer than 20 min, the ice buildup on the evaporator of the two-plate machine insulated the plates and caused the COP to drop off. The lower plate loading on the Remcor reduced the rate of ice buildup on its evaporator plates, enabling it to attain freeze times of over an hour and COPs of 3–3.05. It is evident that both machines achieved the same level of performance despite their different harvesting methods, because the longer freeze times of the Remcor enabled it to overcome the penalty suffered during its harvest cycle. This leads to the conclusion that combining low plate loading with a no-penalty harvest scheme could provide high levels of performance across a much wider range of freezing cycle times.

CYCLIC TESTS-Cyclic part-load tests run on the two-plate machine indicated that for about a 10% load, the COP for cyclic operation was 70% of that for steady-state full-load operation. This level of performance was achieved for compressor run times of 3 min or longer. In general, as the on-off cycling frequency increased, performance of the unit decreased. It was also found that as the on-time of the compressor increased per cycle, performance increased. This leads to the obvious conclusion that the unit should be sized to match its expected load as nearly as possible in order to avoid excessive cyclic operation of the compressor.

FUTURE EFFORTS AND RECOMMENDATIONS-In an effort to reduce the amount of time spent in harvesting by the Remcor, the harvest circuit should be modified to admit the harvesting gas to the top of the plate rather than the bottom. Experience has shown that harvesting from the top down is faster, and therefore would reduce the harvest penalty.

A study of freezing plate performance should be undertaken, using a variable-speed open-type compressor. This would demonstrate the effect of varying plate loading on a given unit and provide information toward determining an optimum combination of plate loading (evaporator area) and capacity (compressor size).

Efforts should be undertaken to simplify the refrigerant circuitry, particularly that part devoted to harvesting. The possibility of coordinating the freeze-harvest cycle of an IMHP with the cyclic operation imposed by a thermostat should be investigated. If harvesting and cyclic operation could be combined in this way, there would be no need for the elaborate harvesting circuits and valving arrangements employed by both units described in this report.

Economic analyses should be combined with component studies to determine the most cost-effective and energy-efficient units for use in space heating and cooling and water heating applications.

The effect of ambient air surrounding the evaporator plates on the ice buildup rate should be investigated thoroughly. As noted in the discussion of ice-making test results, insulation of the freezing plates degraded performance. It will, therefore, be necessary to study IMHP performance, especially harvesting efficiency, with the plates exposed to the cold air of an enclosed storage bin.

REFERENCES

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